

CONSIDERATIONS AND ISSUES IN THE REPLACEMENT OF A CENTRIFUGAL COMPRESSOR TRAIN

by

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INTRODUCTION

The successful justification, design, installation, and commissioning of new major machinery in an existing petrochemical plant requires special consideration of many issues. This is particularly true in instances where existing machines are being replaced with state of the art equipment, which must be integrated into an existing infrastructure. The recognition of the pertinent issues, as well as the strategies used in handling them, directly impacts the overall success of the job.

In this instance, a 20 year old centrifugal compressor and straight condensing steam turbine in the Exxon Baton Rouge Chemical Plant's main ethylene unit were replaced with a new, high performance centrifugal compressor and extraction/condensing steam turbine.

The project is described in four phases. The first, DESIGN INCEPTION AND CONCEPT, contains explanations of how and why the project came about. The second, DESIGN DEVELOPMENT, covers many of the issues considered, including the existing machinery problem history, the installation's physical constraints on the new machine designs, and the logistical considerations of the installation environment and plant operating practices which would affect the new designs. The final compressor, steam turbine, and control system designs are described, as is the factory testing. The third phase, INSTALLATION PLANNING, includes a brief overview of the schedule, with descriptions of how the machines were to be lifted and mounted. The final phase, INSTALLATION EXECUTION AND STARTUP, describes the outcomes of the design and planning efforts covered in the previous phases.

The strategic approaches to the project were key to its evolution and success. These strategies are evident in the text descriptions of the project, with specific examples in the CONCLUSION. They were: 1) awareness of other project or optimization opportunities which could be economically integrated into the main project; 2) comparison of reused equipment performance with the state of the art; 3) testing of past conclusions regarding the viability of changes; 4) investigation of calcula-

tion and assessment assumptions in areas where the resulting conclusions could effect the project's viability; 5) willingness to use new, undemonstrated approaches to difficult problems; 6) intentional configuration of new equipment and systems for similarity to existing ones in order to aid operator understanding; 7) planning to minimize execution dependencies.

DESIGN INCEPTION AND CONCEPT

The Exxon Baton Rouge petrochemical complex consists of a multiproduct line chemical plant adjacent to a large refinery. The current main ethylene unit in the chemical plant was built in 1971-72, and began commercial operation in 1973. Since the initial startup, a number of improvements, debottlenecks and expansions had occurred, and by 1990 the ethylene production capacity was about 60 percent greater than the unit's original design.

In the late 1980s, when the ethylene market became very attractive, plant expansion screenings were conducted and a project was initiated to expand the plant capacity by an additional 10 percent.

Compressor

Increases in the duties of all of the compression equipment in the plant were required for the expansion, but no new streams requiring compression were added. Rerate studies on the existing machinery were conducted early in the design development in order to assess limitations of each machine and best integrate the design parameters for the associated process equipment in each system. In many cases, the rerated system that evolved was compared to a "clean sheet of paper" design, and assessed both operationally and economically. This same sort of evaluation was then made for each individual piece of machinery. The overall goal in these exercises was to be sure that no potential improvements would be overlooked.

The expanded duties could be achieved in all of the compressors by rerating the existing machines. The extent of rerate required ranged from about half the staging to complete new bundles. In all cases, an overriding goal was to use maximum efficiency, state-of-the-art staging, whenever economically justified and physically viable.

The inlet and exit flow path losses of each machine were checked as part of the rerate assessments. In some cases, the losses were excessive by new design standards, but the power lost was not sufficient to justify the corrective action required.

The expansion flow required for the high pressure feed, or charge gas compressor, would be about 10 percent greater than the existing norm, but at the same pressure levels due to other constraints in the process system. Due to the inlet losses, a complete rerate with state of the art staging would be five points less efficient than a new machine from the same manufacturer (77 vs 82 percent). For this machine, operating at 20,000-25,000 hp, there would be sufficient return on the power cost saving to justify a complete, new compressor.

The existing steam turbine driver, a full condensing unit, was sufficient to meet the compressor's needs.

Steam Turbine

About the same time as the ethylene unit expansion process design was completed, a design was being developed to expand the steam systems within the complex to accommodate the expected future needs. One of the main goals of this expansion was to increase the production of steam at 135 psig, the medium pressure level within the complex.

The existing 135 psig system, depicted in Figure 1, was supplied from several heat recovery steam generators (HRSGs), extraction from the steam turbine driving the ethylene plant's propylene refrigeration compressor, exhaust from several smaller steam turbines, imports from the neighboring electric utility plant, and direct letdown from the high pressure (600 psig) steam system. As several of the 135 psig HRSGs are quite old, they will be replaced within the next few years by 600 psig HRSGs, further increasing the need for an efficient means of 600 to 135 psig letdown.

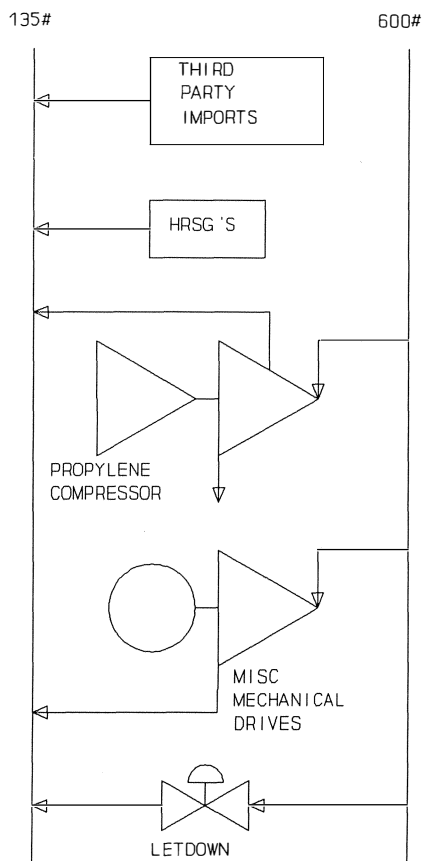


Figure 1. Existing 135 PSIG Steam Producer Block Diagram.

In order to meet this need, the project designers had envisioned the installation of a new, seven MW steam turbine generator (STG). The alternative of installing an extraction/condensing steam turbine to replace the ethylene plant's existing charge gas compressor driver, a straight condensing steam turbine, had not been considered because of past conclusions that it was not feasible to install an extraction machine in the space available. That assumption was "tested" and found to be incorrect for today's machines.

A subsequent scoping study concluded that the cost of replacing the condensing turbine would be about the same as that to install the seven MW STG, with the added benefits of 1) less equipment to operate and support, 2) reduced cooling tower loads for the ethylene plant (less steam to condense) when operating with extraction, 3) greater 600 to 135 psig capacity than the STG, and 4) a new, state-of-the-art turbine and control system vs the existing 20 year old installation. Thus, the decision was made to drop the seven MW STG concept, and progress the design and procurement of a new extraction/condensing turbine for driver of the new charge gas compressor. The future steam system configuration is shown in Figure 2.

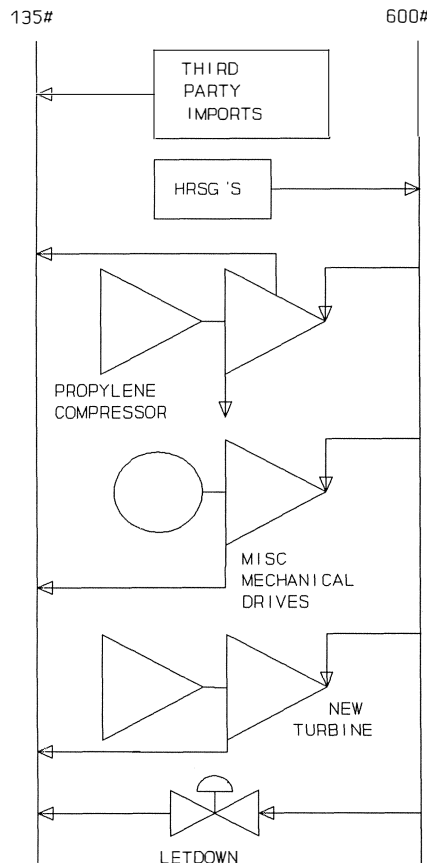


Figure 2. Future 135 PSIG Steam Producer Block Diagram.

DESIGN DEVELOPMENT

For grassroots installations, the design of new machinery is usually dictated only by the process duty required, and industry and customer mechanical specifications. Ancillary systems and components are then designed in order to meet the machine's requirements.

In retrofit cases, however, many ancillaries already exist, the available plot space and installation geometries are limited, there are lists of historical problems to avoid, and operating and control methods are already defined. Further, most retrofits must be executed during relatively short maintenance outages, along side all of the maintenance activities, with the expectation of fully operational systems by the time the rest of the unit is ready for startup. Thus, converse to the grassroots case, in retrofits the machinery must be designed to accommodate the ancillaries. This is depicted in Figure 3.

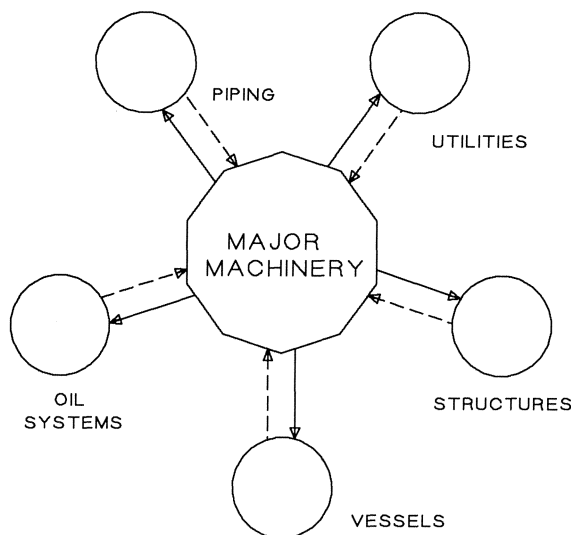


Figure 3. Machinery/Ancillary Design Flow Chart—Grassroots Design Process (solid); Retrofit Design Process (dashed).

Problem History

One of the first considerations in any retrofit design is the problem history of the machines to be replaced, as it is highly undesirable to repeat past problems.

The existing compressor was a horizontally-split, intercooled, two section machine, with the staging arranged inline. The interstage feed gas treatment required that the two sections be segregated, so the “compound” section between the internal volutes of the first section outlet and second section inlet was buffered. The machine’s cross section is depicted in Figure 4.

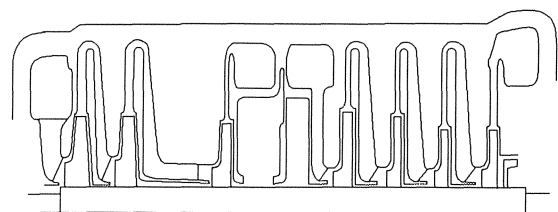


Figure 4. Old Compressor Cross Section.

The compressor operation over the years was generally reliable. Some problems were experienced due to exposures to hydrogen-sulfide and caustic, resulting in severe labyrinth corrosion with subsequent performance deterioration and difficulties maintaining the interstage separation. In addition, unexpectedly high levels of fouling, with accompanying performance deterioration, necessitated the installation of online wash nozzles to the casing within the first few years’ operation.

The steam turbine, (a cross section is shown in Figure 5) was a seven stage, straight condensing unit with a double flow exhaust. The speed control was via a NEMA D oil-relay governor, with overspeed protection using a mechanical bolt. A bar-lift mechanism was used to actuate the six inlet valves. It was mounted with the compressor on a single fabricated baseplate.

The turbine was somewhat less reliable than the compressor. From its initial commissioning solo until it was retired, the machine exhibited high vibration, resulting in a limited allowable operating speed range. This vibration was originally attributed to a resonance of the exhaust housing, although more

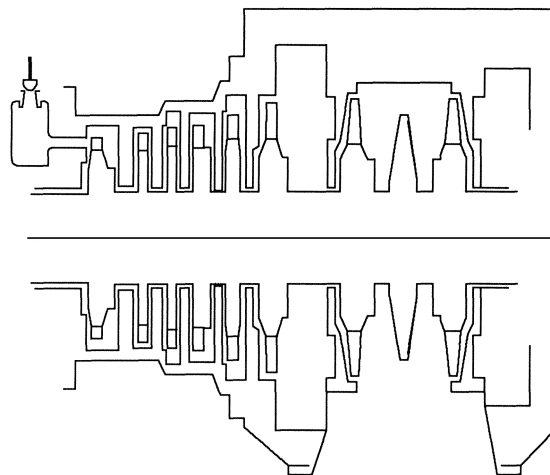


Figure 5. Old Turbine Cross Section.

recent analyses concluded that the vibration was due to the proximity of the second critical.

Physical Constraints

The ethylene, propylene, and charge gas compressors are located in a single, elevated compressor house, having a roof and partial sides. Each train has its own foundation, as does the compressor house structure. A lifting bay on one end of the compressor house is accessed by two overhead bridge cranes. This layout is shown in Figure 6.

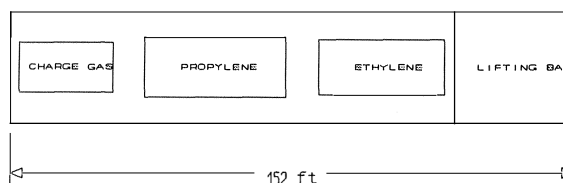


Figure 6. Compressor House Layout.

In order to make the new compressor and steam turbine acceptable projects, the cost of changes to ancillary systems and associated facilities had to be kept as low as possible. Where viable, the new machines were designed to suit the ancillaries. In other instances, compromises were made in assessing the acceptability of those facilities vs the design criteria normally used in grassroots designs, especially where the normal criteria was considered “soft.”

Foundation

The new machines were expected to be bigger and about twice as heavy as the existing ones. Plot space was limited at the time the unit was designed, so it was built “vertically,” with minimum equipment spacing. Major foundation modifications for the new machines were not considered viable. As depicted in Figure 7, virtually the entire area under the machines was taken by piping, the surface condenser, and the condensate pumps. The foundation was also bounded on all sides by oil systems, piping, heat exchangers, or other foundations. Thus, the foundation would limit the size and weight of the new machines.

Both static and dynamic analyses were conducted for the existing foundation. Many of the criteria applied in the design of elevated foundations, such as ratios of mat to super-structure

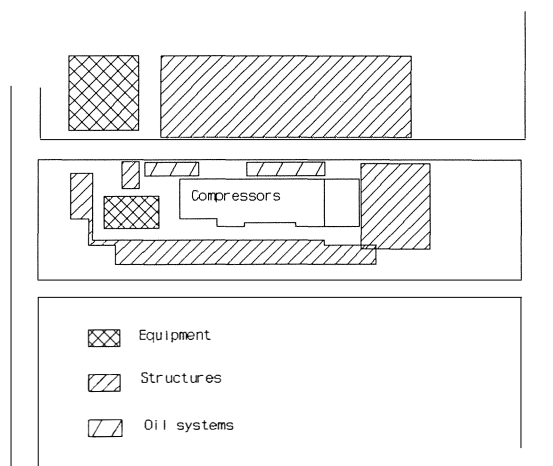


Figure 7. Compressor House Area Layout.

weight and column spacing to elevation, have the effect of stiffening the structure such that its dynamic response is acceptable. Even in the original configuration, the existing foundation did not meet several of these criteria.

Due to the existing turbine's historical vibration problems, several vibration surveys of the structure had been taken over the years. The data from these surveys, conducted with substantial excitations at the turbine, showed negligible responses at the foundation. The absence of any history of concrete cracking or other anomalies in the structure confirmed the lack of foundation excitation.

Machinery foundation dynamic loads are generally assumed to be driven by rotor unbalance forces, which can be considered functions of the rotor weights and speeds. In this instance, neither of the new machine rotors were expected to be as heavy as the existing turbine rotor. The new speed range would be about the same as that of the existing machines. Therefore, since the weight of the heavier rotor would decrease, the speeds of the new machines would be comparable to the existing train, and the foundation was not responsive to the substantial vibration of the existing turbine, foundation dynamic problems were considered unlikely.

However, it was anticipated that a downward shift in the response peaks of the structure could occur because of the heavier machinery. In order to test the significance of this, a finite element (FEA) model of the structure was developed, checked against the results of shaker tests conducted in 1972, then run with the heavier, new machines. Although the validation run did not exactly match the spectrum from the shaker test, the results appeared to reasonably represent the higher-order modes in the running speed range. The response with the new machines was not significantly different from that with the existing machines, although it should be noted that in both runs the locations of the peaks in the running speed range could be significantly impacted by changes in the machinery support stiffness. From this result and the observed historical response of the foundation to machine vibrations, it was concluded that the foundation dynamic response would be satisfactory. The FEA results are plotted in Figure 8.

The foundation static load analysis showed that all structural members would be satisfactory for the new loads. However, the timber friction piles supporting the foundation's mat appeared to be limiting the allowable machinery weights to just over those of the existing equipment.

The original soil reports defined the allowable pile loadings. The static analysis identified the batter piles (piles driven at a

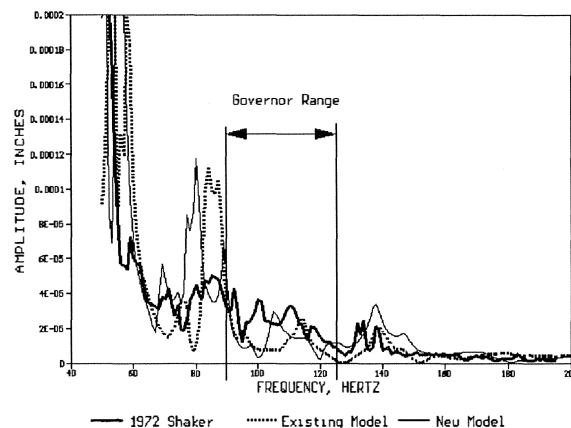


Figure 8. Machinery Foundation Dynamic Response Spectra—Amplitude Per 1000 Lb Force Vs Frequency.

3:12 pitch, designed to take lateral loads) to be limiting. The project's civil engineering consultant recommended that the allowable loading could reasonably be increased by 20 percent because of the conservative, simplified standard calculation methods which included a 2:1 factor of safety.

The foundation lateral loading is generally considered to be due to bending moments resulting from a side load equal to an arbitrary percentage (usually 25-35 percent) of the machine total weight. Actually, any static side loads would be due to piping or wind. Based on API allowable forces and moments, the piping loads would be equivalent to less than six percent of the expected new machine weights. Because of the proximity of other structures, wind was not judged to be a significant load contributor.

Using the original design procedures and 120 percent loading for the batter piles resulted in an allowable machinery train weight of 1.67 times the existing. However, reducing the side loads from 35 to 30 percent increased the allowable new train weight to twice the existing, the target value. The side loads calculated based on the resultants of the allowable flange loadings resulted in batter pile loads equal to only about 56 percent of the allowable. Pile loadings for these three cases plus the original design are shown in Table I.

The static foundation loading was, therefore, concluded to be acceptable for the new machinery string weight.

System Components

In a new installation, the compressor system process vessels would be sized based on a mixture of process requirements and compressor optimizations. In this case, all of the major compressor system suction, interstage, and discharge process vessels were to be reused. This constrained the allowable suction volumes and maximum pressures at each location. In order to meet the required flows and still maintain reasonable safety valve margins, the suction drum velocities were allowed to moderately exceed the established norms.

Several of the compressor OEMs noted that better stage selections, with higher efficiencies, would have been possible if the interstage pressure had been higher. However, the potential gain was not sufficient to make this an economically attractive option, as a significant number of vessel replacements would have been involved.

The new steam turbine was to be designed to operate in a full-condensing mode, if required. Thus, the existing surface condenser and vacuum system would be reused. This set the exhaust centerline location for the new machine, and provided a datum reference for the rest of the train.

Table 1. Foundation Pile Loadings for Various Cases.

	Original 1971	120% Batter pile load	New @ API flg load	New @ 30% side load
WEIGHTS, Klb				
Super-structure	352.8	352.8	352.8	352.8
Machinery	86.2	150.0	172.4	172.4
50% Machinery	43.1	75.0	86.2	86.2
Condenser	103.0	107.0	107.0	107.0
Mat	314.0	314.0	314.0	314.0
Soil	92.0	92.0	92.0	92.0
Piping	50.0	32.0	32.0	32.0
Total Wgt, Klb	1041.1	1122.8	1156.4	1156.4
Number of Piles				
Total, N	30	30	30	30
Trans Batter, Nt	8	8	8	8
Long Batter, Ni	6	6	6	6
Pile Avg Vert	34.7	37.4	38.6	38.6
Mat-mach, H, ft	31.8	31.8	31.8	31.8
Side load %	35	35	N/A	30
Side load, SL, Klb				
transverse	30.2	52.5	8.8	51.7
longitudinal	30.2	52.5	4.4	51.7
Moment, M = HxSL, ft-Klb				
transverse	957.9	1666.9	280.4	1642.1
longitudinal	957.9	1666.9	138.5	1642.1
Pile section modulus, S, ft				
transverse	112.5	112.5	112.5	112.5
longitudinal	178.1	178.1	178.1	178.1
Moment Equiv Pile Load, M/S, Klb				
transverse	8.5	14.8	2.5	14.6
longitudinal	5.4	9.4	1.6	9.2
Avg Total Load, L = P/N + M/S, Klb				
transverse	43.2	52.2	41.0	53.1
longitudinal	40.1	46.8	40.1	47.8
Batter Pile Load, BPL, Klb				
transverse	60.1	80.9	46.9	81.4
longitudinal	62.0	84.2	44.4	84.7
allowable	70.0	84.0	84.0	84.0
% allowable	88.6	100.3	55.8	100.9

1) $BPL = L/\cos(A) + SL/Ni/\sin(A)$; $A = \arctan(\text{pitch})$; $\text{pitch} = 3/12$; $Ni = Nt$ or Nl

2) For API flange load case $SL = F + M/H$; for transverse SL , $F = 8193$, $M = 20248$;
for longitudinal SL , $F = 4042$, $M = 10127$

3) Piping weight for all except Original 1971 case based on actual geometry

The existing combined oil system would also be reused, thus setting the maximum pressures and flows for the lube, seal, and control systems. The lube oil flow was expected to be less than that of the existing machines since the bearings could be lubricated using directed-sprays instead of flooded. The seal oil requirements would not change, as the settling-out conditions

were set by the process interstage safety valves, and as noted above, they would not change. The control oil needs, however, were expected to increase due to the addition of the extraction valve operating cylinder, although it would be viable to increase the pressure, if needed, by removing the control oil pressure regulator. In the final design, the only hardware change was a size increase in the control oil regulator trim to allow the increased flows.

Major Piping

The compressor process piping was several sizes larger than the machine's nozzles in the original construction, swaging down at the machine. Thus, it would be necessary to replace only as much piping as required to reach reasonable termination points, given the revised compressor nozzle locations. Since the existing compressor nozzles were oriented downward in a foundation "cutout," it was possible that the layout of the new compressor's larger nozzles would interfere with the concrete. Therefore, an alternative layout was developed to route the first suction line overhead, if necessary, for top entrance.

The steam turbine extraction piping between the machine and the main plant header would be new (several hundred feet of 24 in pipe), as the existing lateral from the adjacent propylene compressor's turbine would be too small to handle the combined flow. The existing inlet steam lateral (about 150 ft of 12 in pipe) would be replaced for the new turbine, as the maximum throttle flow would be over twice that of the existing machine.

Compressor House

Plot space was limited when the unit was built, so the compressor house layout included very little free space beyond that required for walkways. The overall length was based on the axial dimensions of the three trains, while the width was based on the propylene refrigeration train, the largest of the three compressor frames. Consequently, the new charge gas machines could be somewhat wider than the existing ones, but would have to be about the same axially in order to fit on the foundation and allow for access. This also meant that a vertically split compressor would not be viable, as there would be no deck space available to pull or service the bundle. This is apparent from the charge gas foundation plan, shown in Figure 9.

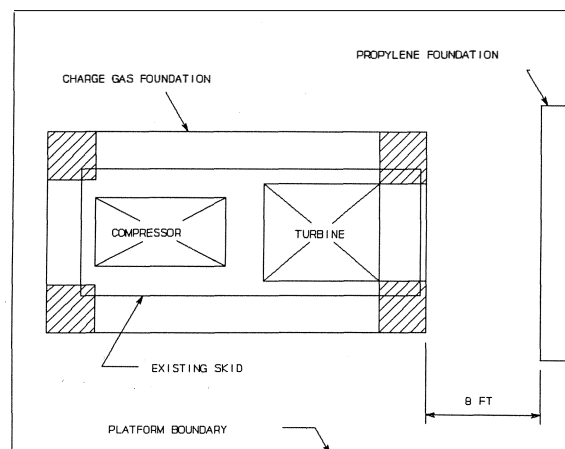


Figure 9. Compressor Foundation Plan.

The structural steel framing of the compressor house was designed in upper and lower frame sections. The upper sections support the roof and overhead cranes on columns common with

the lower sections, which also support the compressor house floor trusses and decking. During the original construction, the lower section was built, the machinery was set using large mobile cranes, then the upper section was built. The upper section frames on the charge gas compressor end were about 15 ft apart, with each frame formed of gable-style I-beams, bolted and welded together. Therefore, a single, common baseplate for the new machines would not be viable, as it would be too large to install through the roof without extensive disassembly, and too long and heavy to lift from the compressor house lifting bay using the overhead cranes.

Logistical Considerations

Controls

Although the new steam turbine for the charge gas compressor would have an electronic governor/control system, it would be configured to mimic the other extraction turbine's controls so that the operators and process engineers would have to deal with only one control logic pattern. The controls on the existing controlled extraction/condensing steam turbine driving the propylene compressor consisted of a mechanical oil-relay governor and three-arm linkage. Control signals from the process control computer were sent to the governor to adjust the speed to hold the compressor suction pressure setpoint, and to an extraction linkage actuator to hold the extraction flow setpoint. Since the new turbine's control system would be microprocessor based, programming to mimic the other turbine's controls would be easily accomplished. Both sets of controls are schematically depicted in Figure 10.

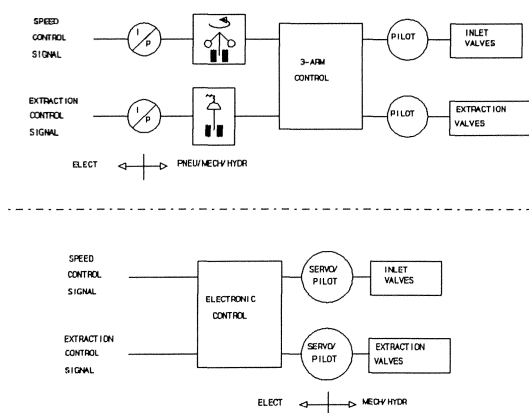


Figure 10. Turbine Control System Block Diagrams—Existing (top) and New (bottom).

Execution Environment

The existing machines would be removed and the new ones installed during a unit major turnaround maintenance outage, slightly extended due to the project work. Due to the extent of product coordination required and downtime costs for this type of outage, the incentives for a timely, troublefree startup would be substantial. Thus, there would be no time for "shakedown" runs, and no tolerance for startup problems.

During past turnarounds, the machinery work had set the "critical path" and constituted the highest priority activity. However, during this outage the nonmachinery work would be far more extensive than normal, setting the critical path and having priority over the machinery work where conflicts existed. In addition, the machinery work would be planned and directed by inhouse resources, while a general contractor would

be responsible for the rest of the work on the unit, including the machinery-related instrument and electrical, piping, and structural activities. Logistical support for the machinery work, such as mobile cranes, would also be supplied by the general contractor. With the general contractor's priority being on nonmachinery activities, yet supplying logistical support for the machinery work, it was prudent to plan the machinery work such that dependencies on the general contractor would be minimized.

Because of the unit's minimum equipment spacing and "vertical" construction, ground-level congestion was expected to be a problem, with equipment laydown and crane spotting space at a premium. This made it preferable to use the compressor house overhead cranes for the new machine lifts vs a large, mobile crane.

Machinery Designs

The machinery designs were based on API 612, 617, 670, and 671, in addition to the company applicable specifications. Since these would be new machines, state-of-the-art performance was expected.

An individual, column-mount baseplate was specified for each machine. Individual bases were chosen in order to make the installation lifts light enough for the compressor house overhead cranes, or small enough to fit through the compressor house roof if a mobile crane were used. Column-mount bases were specified in order to save time on the installations by using adjustable wedge machinery mounts at the "point-support" locations, rather than full-support grouting.

Compressor

The new compressor, whose cross section is shown in Figure 11, was specified as a horizontally split machine in a back-to-back configuration, vs the existing machine's inline configuration. This eliminated the need for an interstage separation labyrinth, as the balance piston leakage would serve that function. The machine was also shorter by several inches in the back-to-back configuration, since the balance piston could be located in the same axial space as the interstage volutes.

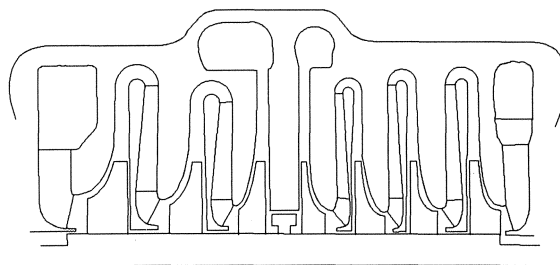


Figure 11. New Compressor Cross Section.

Since energy saving was the basis for the project to replace the existing compressor vs a rerate, an ASME PTC-10 class 3 factory performance test was required. A power tolerance of one percent was specified vs the API 617 tolerance of four percent, as a four percent deviation would have consumed most of the credits figured into the project's justification.

The flow path labyrinths were specified as polymer rub strips with rotating knife-edges in order to eliminate potential corrosion problems while maintaining the clearances necessary for state-of-the-art performance.

Online wash nozzles were specified at each stage return channel. Unlike those on the existing compressor, the new nozzles would not be removable on-the-run, as the packing

gland necessary to allow retraction was a source of fugitive emissions, and the safety of online removal was a concern. Further, there was no history of clogged wash nozzles on the existing machine. In the one instance when a nozzle was removed online for inspection, nothing was found.

The new compressor was considered susceptible to rotor-bearing system self-excited instability, as the gas densities would be sufficiently high to generate appreciable excitation, and the balance piston would be near the rotor's midspan. In order to mitigate this phenomenon, a balance piston gas injection swirl break was incorporated into the design, along with shaft-end seal assemblies incorporating squeeze-film damper rings. For verification of the effectiveness of these features, the OEM proposed factory testing to quantify the system's damping ratio. This test would be conducted with the machine loaded, after the performance test.

Steam Turbine

Due to the timing of the steam system expansion design effort, the new compressor specification had been completed well before the new steam turbine was even considered. In fact, compressor proposals were being prepared by the prospective suppliers by the time the decision was made to progress the new steam turbine.

The turbine replacement was not expected to have a large impact on the compressor design, since the speed range was set by the compressor frame size, which was limited by the foundation and platform size. Thus, a separate specification was prepared for the new steam turbine so that the each machine selection could be evaluated independently in order to optimize the string.

The new charge gas compressor's turbine would be configured similar to the adjacent turbine driving the propylene compressor, with a controlled extraction.

In order to facilitate the rapid shedding of high pressure steam consumption in the event one of the plant's steam supplies were lost, the turbine design was specified to achieve maximum flexibility in the operation of the extraction, with the ability to produce design output anywhere between zero extraction and cooling flow only to condensing. This meant that both sections of the turbine would be large, with the high pressure section sized like a back-pressure machine, and the low pressure section sized to pass full condensing flow in the zero extraction mode. The peak efficiency of the turbine would be slightly compromised as a result, but in the full condensing mode, it would still be as efficient as the old turbine. The new turbine cross section is shown in Figure 12, while a typical maximum flexibility performance map is shown in Figure 13.

In the initial case of only the compressor being replaced, the existing coupling would be reused to the extent possible. However, with both the machines being replaced, coupling replacement would be cost effective, and adding torque monitoring would be viable. Power monitoring would be used by the process engineers to optimize operations, and could also validate the efficiencies claimed in the project economics. Thus, a torque-meter coupling was included as part of the new turbine's specification.

Since the turbine was being replaced, there would be an opportunity to conduct a complete factory string test. Although it was not considered necessary, string testing would be desirable in order to validate the torque-meter and demonstrate the train's mechanical integrity. The final decision regarding a string test depended on the supplier selections, logistics, and costs, but the ultimate selection of a single supplier for both machines made string testing possible with no logistical complications and no extra cost.

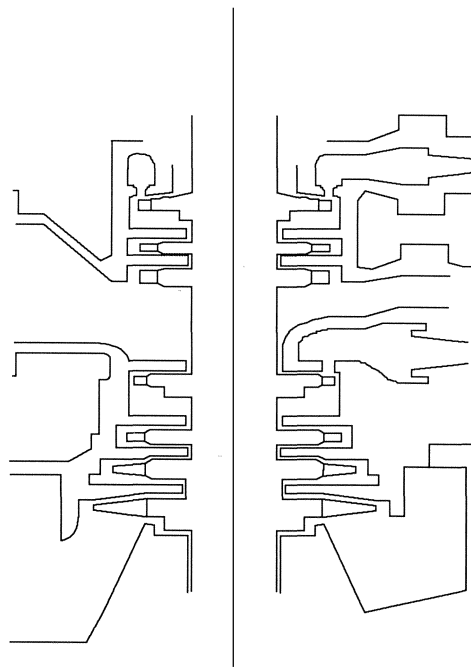


Figure 12. New Turbine Cross Section.

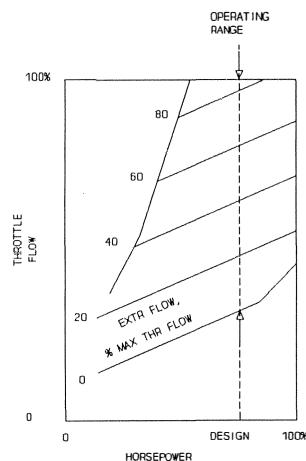


Figure 13. Typical Maximum Flexibility Steam Turbine Performance Map.

Controls

A triple, modular, redundant (TMR), microprocessor based governing and control system, with dual coil servos, was specified for the new turbine. The original concept was that only the turbine speed and extraction would be controlled from the new system, using control inputs from the process control computer. As the design evolved, it became apparent that the new system could also be used for trip logic determination, so the project was slightly expanded to add the sensors needed to allow two-of-three trip logic. Where practical, transmitter signals were used instead of switch contacts in order to provide "live signals," allowing continuous monitoring by the TMR system for deviant or faulty inputs.

An existing 120 VAC uninterrupted power supply (UPS) system provided one power source for the new control system, while a 125 VDC UPS system would be installed to provide a

second power source. The trip output signal from the TMR system would simultaneously actuate two solenoid operated valves (SOVs) to depressure the control oil loop, with one SOV powered from each of the UPS systems. The SOVs would be piped such that actuation of only one of the two would be required to dump control oil.

The control oil lines at both SOVs and the trip-throttle valve (TTV) would include valving to allow periodic online testing. A manual chock would be provided to prevent the TTV from closing during its test. Each test valve would be instrumented with a position switch for input to the control system to alarm and inhibit testing if the valve lineup were incorrect.

As is typical with extraction steam turbines, the allowable stresses in the new machine's last high pressure stage would dictate minimum allowable extraction pressures. The adjacent turbine used a control valve in its extraction line in order to limit the minimum extraction pressure. In order to reduce costs, the analogous valve was left out of the new turbine's extraction in favor of a low extraction pressure trip to protect the machine. The allowable pressure vs throttle flow curve was programmed into the TMR control system as an override to the extraction flow setpoint. This logic would drive the turbine to full condensing in the limiting case. While this would be undesirable for the steam system, the process operation would be sustainable, as there would be adequate margin between the historically observed minimum medium level steam pressure and the allowable extraction pressure.

The operator interface panel, shown in Figure 14, was made as simple as possible in order to minimize the potential for operator errors. Push-button switches would initiate each phase of the startup sequence, switch speed and extraction control to either local or process computer control, and allow raising or lowering of the speed or extraction settings during local control. A keyboard and CRT screen would be used for data access, but these would be used primarily for troubleshooting and would not be required to start or operate the machines.

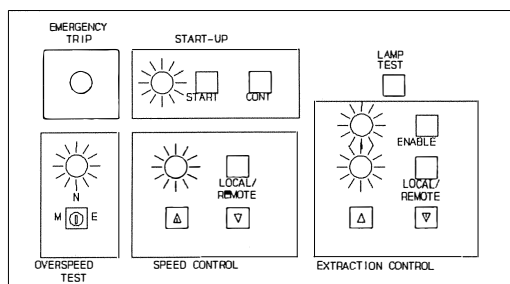


Figure 14. Operator Local Control Panel.

The new control and UPS systems would be housed in a new instrument shelter to be situated on an extension of the compressor deck. The shelter would be sized for future relocations of existing machinery monitoring and vibration systems housed in other places but connected to the new system. Space was also left for a future TMR system for the propylene compressor train.

Machinery Testing

Factory testing was conducted for the new machinery, with mechanical runs for all rotors, unbalance response tests, a compressor performance test, and a compressor stability test. The new control system received a factory acceptance test, but in order to minimize the potential for shipping damage, was not used in the machinery tests.

All of the machinery tests were conducted with the machines mounted on their baseplates. Because the bases were designed for column mounting, they were over five times more rigid than the alternate full support designs would have been. Thus, changes in the vibration characteristics due to differences between the factory and field support characteristics were unlikely.

No problems were encountered in any of the tests. The demonstrated vibration response during each unbalance response test was as predicted. The compressor's PTC-10 equivalent gas performance test revealed slightly better results than predicted over a wide flow range. This demonstrated compliance with the contract specification of one percent power tolerance. The project economics were also satisfied, as the tested efficiency was 85 percent at the design point, three points better than the project justification basis.

The compressor stability test showed damping ratios slightly better than predicted, and confirmed the credibility of the calculation methods and stability predictions at the design operating conditions.

INSTALLATION PLANNING

Manning and Timing

The total installation time was expected to take 47 calendar days, with the machinery portion manned with six people working six ten-hour shifts per week. The removal of the old machines, foundation and structural preparation, and setting the new machines was planned at 21 days. To the extent possible, the new piping and conduits were to be installed preshutdown or during these first 21 days, with the final tieins, steam blows, and oil flushing to take place during the following 14 days. Instrument loop checks, the turbine solo, and coupling installation were to be done during the remaining time.

Floor and Foundation

The new machines and bases were about twice as heavy as the existing compressor and turbine, but as discussed earlier, were still within the load capacity of the foundation. The new bases were designed to reuse the existing anchor bolts, but because of the added height of the adjustable wedge supports, special extended nuts were designed to reach the anchor bolts. One new cored, epoxy adhesive-mounted anchor bolt was required under the turbine gib key. An extension was designed to add stretch length and allow tightening access for this bolt from the top of the turbine skid.

The new machines were about the same axial length as the old ones, but were somewhat wider. The bases would still fit on the foundation with adequate walkways on each side, but modifications to enlarge the compressor house floor cutout and structural steel were required. The new machines were also taller because of the column mount bases and turbine valve gear, but were still within the capability of the overhead crane's head space for maintenance lifts.

The planning basis was to remove the old machines, then remove the old baseplate, breaking out the existing epoxy grout as needed. If necessary, a new grout cap would be poured to seal any discontinuities. The adjustable wedge supports would be set and leveled, and individual grout pads would be poured under each support. The new machines would be lifted, set, then leveled with the adjustable supports.

Lifting

Significant work was required in order to develop the lifting concept for the new machines. Each machine was designed with an individual base in order to be small enough to fit in the compressor house lifting bay and light enough to lift with the compressor house overhead cranes. Alternatively, a large mo-

bile crane could be used, lowering the machines thru the compressor house roof without disassembling any significant structural members.

The two compressor house overhead cranes were of different capacities, with the smaller one rated at just over 50 percent of the larger one's capacity. The larger crane was part of the original installation, while the smaller crane had been added later to expedite maintenance turnaround work.

The initial lifting concept was to use only the larger compressor house overhead crane. The project's civil/structural engineering consultant had reviewed the compressor house and foundation, and concluded that the required lifts were acceptable, although one of the column-line pile groups would be loaded beyond its design capacity. The crane manufacturer advised that the crane could be uprated slightly, then be suitable for a 125 percent "planned engineered" lift, per ANSI B30.2. However, the inplant safety committee was very reluctant to agree to any "over-rating" lifts, because although OSHA 1926 references and quotes from ANSI B30.2, it makes no specific references to allowable lifts over a crane's nameplate rating.

The use of a mobile crane was investigated, then dismissed because of logistical complications. The only place available to spot a crane of the required capacity would have blocked access to the compressor house lifting bay. About five days setup and teardown would have been required for the crane under consideration, further restricting both machinery and nonmachinery work in that area. In addition, the crane rental fees would have been substantial. Thus, the mobile crane was considered to be a "last resort" option.

The lifts were ultimately planned using both compressor house cranes. Followup studies of the crane support structure were conducted for the two crane lifts in order to assess the effects both the added weight of the second crane and the redistribution of the load along the crane rails. The foundation pile groups at the column line identified by the civil/structural consultant were the only area of concern, being about 50 percent overloaded.

Subsequent analyses and comparisons with the design calculations for the original construction showed that the consultant's assumed dead loads for the structure were excessive. In addition, the actual width of the new machines would limit the possible centerline offset of the trolleys, resulting in greater load sharing between columns than would result with the trolleys at their travel stops. As a result, the structural loadings for the lifts of the new equipment using both cranes would be acceptable, and would actually be less than the loadings in the original design calculations for the heaviest maintenance lift.

The correct hook locations for the two cranes relative to the each load's center of gravity were determined based on the minimum distance between the cranes and the load distribution required to keep the hook loads at approximately equal percentages of the rated lifting capacities. However, the lifting lugs on the new compressor and turbine bases were not positioned in the required locations, and the bases were actually shorter than the distance between the crane hooks. Therefore, a lifting frame was designed, built, and load tested. Since the lift bay was on the opposite end of the compressor house from the charge gas train, the amount of head space available was critical. With the other machines stripped to their splitlines for their overhauls and rerates, the lifting frame reduced the head space requirement such that the vertical clearance was adequate. The turbine lift plan, showing the layout and rigging requirements, is depicted in Figure 15.

A remaining issue with the two crane lift was that the compressor would still be about two feet from its required location

LIFT DATA FOR 38 TON OVERHEAD CRANE	
64.5 % OF TURBINE WEIGHT	55,470 LBS.
(2) 35 TON SHACKLES WEIGHT	93 LBS.
(2) 25 TON SHACKLES WEIGHT	60 LBS.
EST. WT. OF SON OF LIFT BEAM	4,888 LBS.
(2) 55 TON SHACKLES WEIGHT	180 LBS.
(2) 2" X 5" SLINGS WEIGHT	210 LBS.
TOTAL WEIGHT LIFTED	60,881 LBS.
OVERHEAD CRANE CAPACITY	76,000 LBS.
PERCENT OF CHART	80 %

LIFT FOR 20 TON OVERHEAD CRANE	
35.5 % OF TURBINE WEIGHT	30,530 LBS.
(2) 35 TON SHACKLES WEIGHT	93 LBS.
(2) 25 TON SHACKLES WEIGHT	60 LBS.
EST. WT. OF SON OF LIFT BEAM	3,982 LBS.
(2) 55 TON SHACKLES WEIGHT	180 LBS.
(2) 1.5" X 5" SLINGS WEIGHT	110 LBS.
TOTAL WEIGHT LIFTED	34,955 LBS.
OVERHEAD CRANE CAPACITY	46,000 LBS.
PERCENT OF CHART	87 %

RIGGING DATA FOR TURBINE LIFT	
(4) 35 TON SHACKLES	
TOTAL CAPACITY 280,000 LBS.	
TOTAL WEIGHT LIFTED 86,000 LBS.	
PERCENT OF CAPACITY 30 %	
(4) 25 TON SHACKLES	
TOTAL CAPACITY 200,000 LBS.	
TOTAL WEIGHT LIFTED 86,000 LBS.	
PERCENT OF CAPACITY 42 %	
(4) 55 TON SHACKLES	
CAPACITY PER SHACKLE 110,000 LBS.	
MAXIMUM TENSION AT 40 DEGREE ANGLE PER SHACKLE 46,262 LBS.	
PERCENT OF CHART PER SHACKLE 43 %	
(2) 2" X 5" SLINGS	
CAPACITY PER SLING 62,000 LBS.	
MAXIMUM TENSION AT 40 DEGREE ANGLE PER SLING 46,262 LBS.	
PERCENT OF CHART PER SLING 75 %	
(2) 1.5" X 5" SLINGS	
CAPACITY PER SLING 38,000 LBS.	
MAXIMUM TENSION AT 40 DEGREE ANGLE PER SLING 26,561 LBS.	
PERCENT OF CHART PER SLING 74 %	

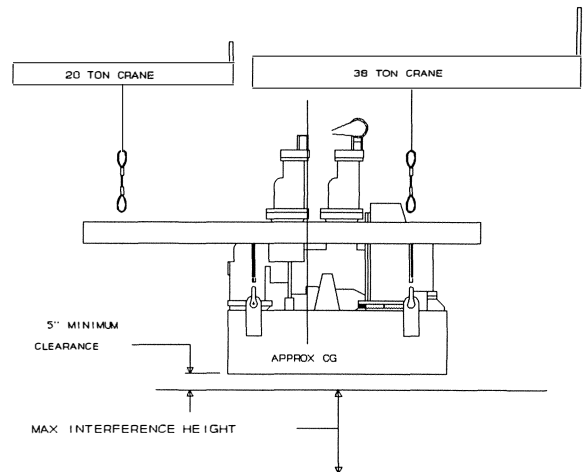


Figure 15. New Turbine Lift Plan.

when the cranes were at the end of their rails. The solution to this was to lower the compressor base onto rollers sitting in steel channels on the foundation, roll the machine to the required location, then use jacks to remove the rollers and lower the base onto its supports.

INSTALLATION EXECUTION AND STARTUP

The new control system shelter and as much of the new conduit and piping as possible were installed prior to the unit shutdown. Many components that could not be installed until the shutdown were prefabricated and strategically located to facilitate installation.

The removal of the old machines proceeded as planned and was several days ahead of schedule. When the old baseplate was removed, the old grout appeared to be in good condition. How-

ever, due to miscommunication, the old grout and several inches of foundation concrete were broken out vs simply roughing up, for the grout cap as planned. This necessitated getting more grout and making larger pours than planned, at a cost of several days.

The lifts of the new machines, shown in Figures 16 and 17, occurred exactly as planned. However, the progress was impeded when the cast iron housings of several of the adjustable wedge machine supports broke due to over adjusting the wedges. Final leveling was ultimately accomplished with a combination of the wedges and solid shims. In another deviation from the original plan, grout blocks were poured at each support location, encapsulating the supports and shims, and providing more conventional supports for the machines.

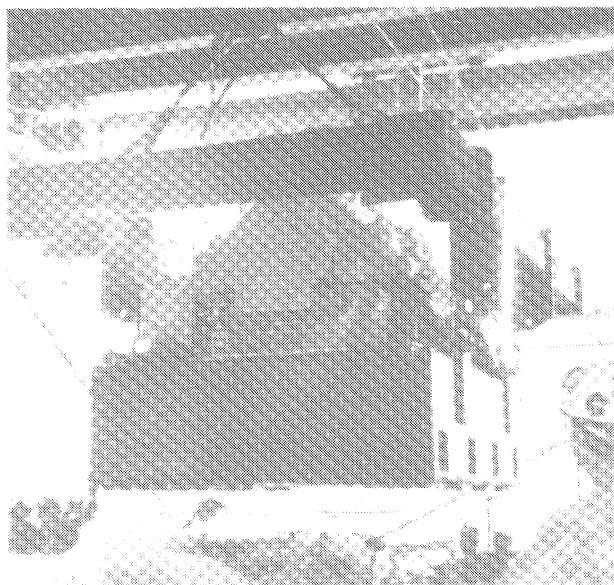


Figure 16. New Compressor Lift.

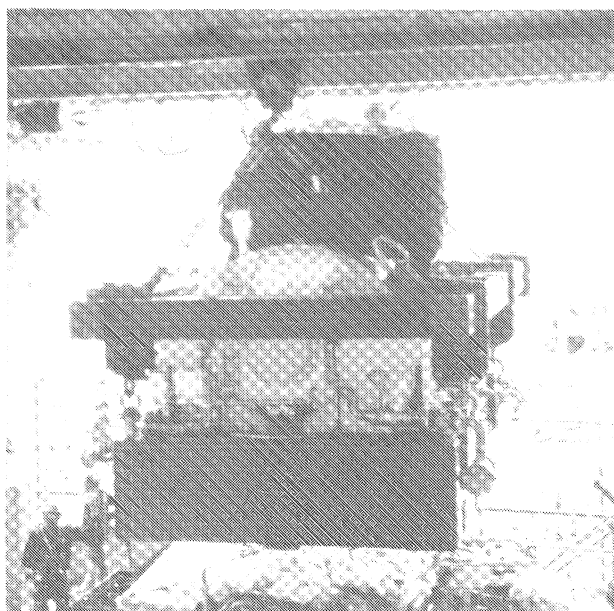


Figure 17. New Turbine Lift.

The machines were turned over to the pipefitters and instrument technicians on schedule, with the time lost due to the grouting about equal to the time gained during removal of the old machines. However, due to insufficient manning, rework of prefabricated piping, difficulties with major piping fit-ups, and delays on the nonmachinery work, the job fell behind schedule (as did the rest of the unit work), ultimately requiring much longer than planned (77 vs 47 days).

Later in the job, the major steam piping blows and oil system flushing were executed as planned, albeit behind schedule. The control system checkout and turbine solo also went according to plan, with very few anomalies. At the startup, the machinery and controls functioned smoothly with only a few minor control system programming issues, all corrected online.

From the operation to date, all systems are functioning as expected, with both the turbine and compressor slightly exceeding the expectations of their design and test stand performance.

CONCLUSION

The replacement of the ethylene plant's charge gas compressor train has been briefly described. The success of the job was attributable to many factors, but there were several key aspects in the approaches used that ought to be highlighted. These could apply to any job, but are especially applicable to retrofits, where there can be unique "boundary conditions."

It is important to maintain contact with project or longterm planning groups in order to be aware of opportunities. The originally planned seven MW STG installation was not dependent on the ethylene plant downtime, with the STG planned to startup about six months later. Even if the steam turbine replacement would have been identified as a viable alternate during the STG project development, there would have been insufficient lead time to develop and progress it for the installation window. It was only by timely knowledge of the existence of the STG project that the turbine replacement was conceived.

Knowledge of the state of the art is essential to assess the value of upgrades and avoid unwittingly bypassing opportunities. It was only by checking the existing compressor's rated performance vs that of a new, state of the art machine that the compressor replacement opportunity was identified. The incremental addition of the train shutdown controls to the turbine control system became a viable upgrade step once the decision was made to use a TMR system.

Testing the past conclusions that an extraction turbine could not fit in the existing condensing steam turbine's place, and finding those conclusions to be invalid, resulted in an installation previously considered impossible.

Understanding the assumptions and bases built into the results of the existing foundation capacity and crane support structure studies allowed more thorough analyses to be made. The initial marginal or unacceptable results were ultimately invalidated, and the project progressed.

Creative approaches were applied to a number of issues. Some, like the alternative of a top-entering compressor main inlet connection, allowed the consideration of more options in the search for the best combination, but ultimately were not used. Most of those actually planned and executed, like the new machine lifts with the lifting frame and overhead cranes, were highly successful. A few, such as the use of the adjustable wedge machine supports with minimal grouting, had to be modified in order to achieve a successful outcome.

An awareness of the need for "operator friendliness" was maintained throughout the project, with the most apparent examples being the simplicity of the operator's panel, and the control set-up analogous to the adjacent extraction turbine.

Almost from the beginning, the new machines were conceived such that dependencies on other resources would be minimized during the installations, with the best example being the lifting plans for the new machines. This proved to be the best approach, as the aspects of the installations which went most awry, and most impacted the schedule, were those under the control of the general contractor.

